| Load case | Modulus direction | Detailed fins deflection | Solid equiv fin Deflection | modulus | ratio | New modulus |
|---|---|---|---|---|---|---|
| Shear along length | Gyz | 0.01894 | 0.011223 | 293.58 | 0.5926 | 173.96 |
| Shear across width | Gxz | 0.0015098 | 0.00058 | 5962.50 | 0.3841 | 2290.14 |
| Vertical Z tension | Ez | 0.0001073 | 0.0000092 | 361874.45 | 0.0854 | 30898.33 |
| | | Reaction | Reaction | | | |
| Tension width (fin only) | Ex | 835.71 | 836.8 | 253772.58 | 0.9987 | 253442.02 |
| Tension length (fin only) | Ey | -0.0048317 | -0.0048317 | 1.55 | 1.0000 | 1.55 |
| Shear xy (fin only) | Gxy | -10.767 | -10.767 | 328.42 | 1.0000 | 328.42 |

*Table1 - Equivalent stiffness calculation – example iteration*

| Load case | Modulus direction | Detailed fins deflection | Solid equiv fin Deflection | modulus | ratio | New modulus |
|---|---|---|---|---|---|---|
| Shear along length | Gyz | 0.01894 | 0.01894 | 174.25 | 1.0000 | 174.25 |
| Shear across width | Gxz | 0.0015098 | 0.00152 | 2426.35 | 1.0045 | 2437.28 |
| Vertical Z tension | Ez | 0.0001073 | 0.0001066 | 31082.69 | 0.9936 | 30882.74 |
| | | Reaction lb | Reaction lb | | | |
| Tension width (fin only) | Ex | 835.71 | 835.57 | 253772.58 | 1.0002 | 253815.10 |
| Tension length (fin only) | Ey | -0.0048317 | -0.0048317 | 1.55 | 1.0000 | 1.55 |
| Shear xy (fin only) | Gxy | -10.767 | -10.767 | 328.42 | 1.0000 | 328.42 |

*Table 2 - Equivalent stiffness calculation – final solution*

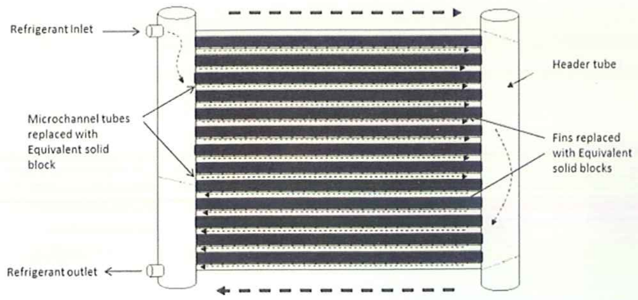


*Figure 5 - Microchannel assembly with equivalent solid blocks*

### Validation

This methodology was validated using a light commercial HVAC unit for a thermal stress case by performing tests in the laboratory. Strain gages were placed on the microchannel tubes near the header, a high stress location, and the thermal strains were measured. The test results closely matched with that of the simulation, deviation being close to 10%. This validation provides good confidence on the equivalent modeling methodology.

### Modal and Thermal Analysis of Complete Assembly

### Modal Analysis

An analysis was undertaken for a transport HVAC unit, where dynamic response due to road vibration input is an important consideration. The equivalent model with tuned mass and stiffness is used to build the complete condenser coil which is used in the unit. The modal response of the coil and its influence on the supporting structure is as shown in Figure 6. These modes are dominated by bending about the weak axis of the coil slab, which corresponds to the strong axis bending of each tube. The strong axis equivalent modulus for the tube was used in this case.

### Thermal Analysis

Microchannel coils are subjected to temperature gradients spatially, due to the heat transfer from the fluid inside. In a multi-pass condenser coil, the hot entering fluid is very close to the cold leaving fluid, which creates significant thermal stresses. These stresses can lead to catastrophic failure of the coil or of the supporting structure. Hence evaluation of thermal stresses becomes critical in order to have a reliable system in the field. The temperature distribution across the coil is applied as a load, and a steady state thermal analysis is peformed. The thermal profile after steady state analysis and stress generated due to temperature is for the transport HVAC unit studied.

### Conclusions

The new method developed to build an equivalent model of the condenser coil using equivalent stiffness solids was found to have close behavior to that of the actual model. This method overcomes the challenges faced by the analyst in capturing the complex geometry of the microchannel. The model developed has comparatively low number of elements, and a good control on the mesh quality can be established. Investment in terms of time and system resource is largely reduced.

The use of this methodology in developing the condenser coil or any similar components/structures can be validated by comparison of complete system response against test results. The modal and thermal response obtained for the complete assembly by the use of simplified model captures the behavior of system in application. Hence the new methodology is beneficial in modeling the complex system without compromising the accuracy of the results.

### Future Scope

Validation of this methodology with actual prototype build of this transportation unit has been planned to include both thermal and dynamic tests. The outcome of these tests will provide much more confidence in the methodology.


*Raghu Kumar G, Gregory Meeuwsen*
*Ingersoll Rand*


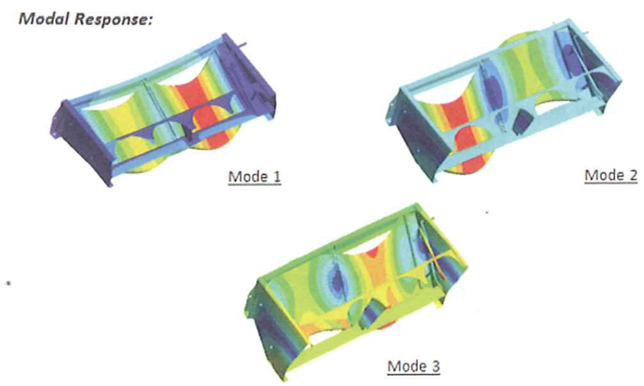


*Figure 6 - Modal response of the unit inclusive of the condenser coil*



# FE modeling of a composite sandwich laminate with LS-DYNA for Aerospace applications

Laminated composite materials such as sandwich structures are widely used in the transportation industry due to their high stiffness and strength to weight ratio. Significant efforts have been made in improving the structural theories and prediction methods to give engineers a reliable tool to develop new structural concepts (i.e. vehicle front and rear end components, automotive and aeronautical seats). The complete mechanical behavior of a composite material is very difficult to be predicted without first calibrating the model to some test data. This is especially important when damage and rupture are considered for impact and crash analyses.

This project investigates how best to correctly predict the energy absorption of these non-isotropic and non-homogeneous materials, whose lamina thickness is typically of the order of 0.2 mm or less and consists of tiny fibers or particles dispersed in a matrix material. This makes it impossible to model details in the micro scale when considering a whole component or vehicle. The typical FE element size for metal structures is 5mm and needs to be similar for composite structures to ensure manageable simulation times.

Therefore, in order to achieve computational efficiency relevant models and theories need to be understood. This article examines some modelling techniques to simulate a sandwich composite laminate, paying particular attention to its non-linear behavior and correlated energy absorption features. The numerical results have been compared with experimental tests.

### The composite sandwich laminate

For this study, 3 samples of composite sandwich laminates have been kindly provided by Benecor Inc. The sandwich laminate samples have carbon fiber composite faces and a core made of commercially pure titanium (TiCP). The faces are made of 9 unidirectional fiber plies in an epoxy matrix, with a [0°90°0°90°0°90°0°90°0°] ply book. The core is made of square cells welded at the intersection corners. A sample of the provided sandwich is shown in figure 1.

*Figure 1 - Composite sandwich laminate object of the study*

|  | mm | Inches |
|---|---|---|
| Width | 50,80 | 2,0 |
| Length | 152,40 | 6,0 |
| Total height | 12,70 | 0,50 |
| Face thickness | 1,15 | 0,045 |
| Core height | 10,40 | 0,41 |
| Cell side | 3,810 | 0,150 |
| Cell thickness | 0,0762 | 0,003 |

*Table1 - Geometrical sample sizes of the laminate*

*Figure 2 - Shell-brick-shell model of the sample of the sandwich laminate*

The geometrical properties of the sandwich samples are summarized in table 1. The density of the TiCP core is 11.7 pcf (187.4 Kg/m3).
The mechanical properties of the materials (carbon-epoxy lamina and CP titanium) were not available.
Reasonable values for the first analyses were obtained from literature and then a combination of experimental tests and trial FE analyses were used to obtain more reliable material data.

### Mathematical modeling strategy

Many models are available that simulate the mechanical behavior of a sandwich laminate: their degree of complexity and accuracy depends on what behavior the model was designed to assess. In general, it is advisable to use the simplest possible model to predict the desired phenomena.
For example, if the aim is to predict linear stiffness and displacements, critical buckling load or early natural modes, a FSDT plate model that accounts for a correct shear factor is usually good enough. If other phenomena have to be investigated, such as the correct stress and strain field of each lamina, further eigenvalues and eigenmodes, or if the structure can be subjected to large non-linear deformations that could lead to degradation and failure, a refined solid model or a higher order plate model is recommended.

For this study, the aim is to assess the energy absorption of the laminate, therefore a good compromise between model detail / accuracy and simulation time is a shell-brick-shell model (the core modeled as a solid between two shell layers). The upper and lower layers can be effectively modeled with an equivalent FSDT plate or a higher order theory, with possibly stress recovery for augmented accuracy.
The main advantage of this modelling method is that it can predict better the kinematic behavior under non-linear combined loads, which include large and plastic deformation or stresses and strains normal to the mid-plane. The shell-brick-shell model can also be used to model failure modes (that can be many and complex for a sandwich laminate), such as skin failures (e.g. face yielding or face wrinkling), core failures (shear or indentation), or the adhesive failure between the core and outer layers. This can be done using a TIE _ BREAK contact definition in LS-DYNA, but has not been studied in this paper. All these features are important in correctly predicting the energy absorption of the sandwich laminate.

### The model for the skins

One important decision is to choose the right material model for the skin laminate. LS-DYNA has a wide range of material models, with a number of materials specific for composite laminates.
For nonlinear analyses, it is important for the materials to include plasticity, stiffness degradation and failure. For dynamic analyses, it is also advisable to use a correct value of damping. All these properties influence the energy absorption of the structure, and are therefore of utmost importance in explicit FE crash simulations.
The relevant material models for composite shells in LS-DYNA are MAT 22, 54, 55, 58, 59, 104, 108, 114, 116, 117, 118, 158, 161, 162. The large number of materials shows the huge effort made to create models that satisfactorily predict the behavior of laminated composites. Figure 3 shows the behavior of some of these materials.

In addition to the LS-DYNA manuals, it is always helpful to refer to relevant technical papers, run some benchmark simulations, and compare the results with hand calculations or known results. This study incorporates all these methodological aspects for a sandwich laminate.
For this study, MAT 54 has been selected as the modelling material for sandwich skins, since it appears to be the best compromise

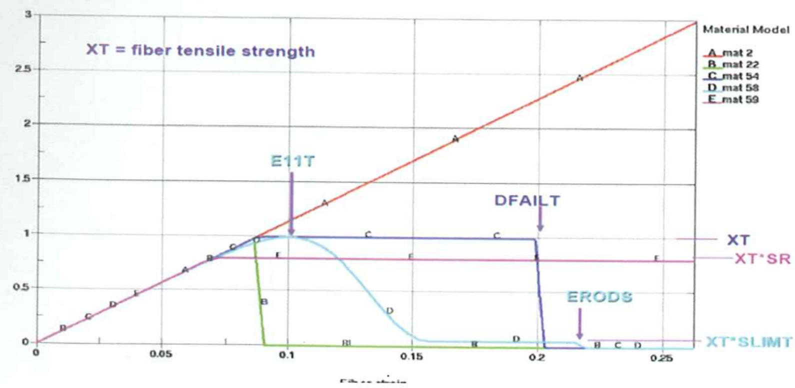


*Figure 3 - Constitutive laws for some popular materials in LS-DYNA [LSTC Corporation, Modeling of Composites in LS-DYNA, internet source]*

| | mm | Inches |
|---|---|---|
| Width | 50,80 | 2,0 |
| Length | 152,40 | 6,0 |
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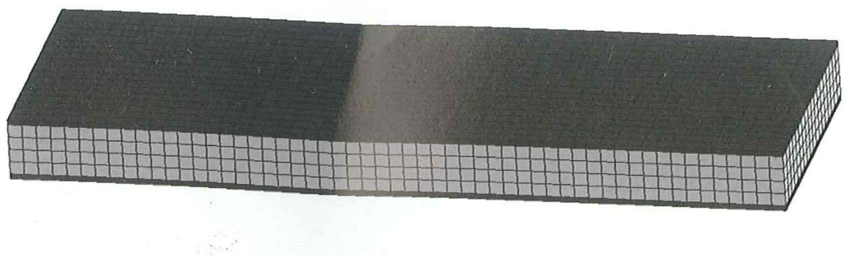

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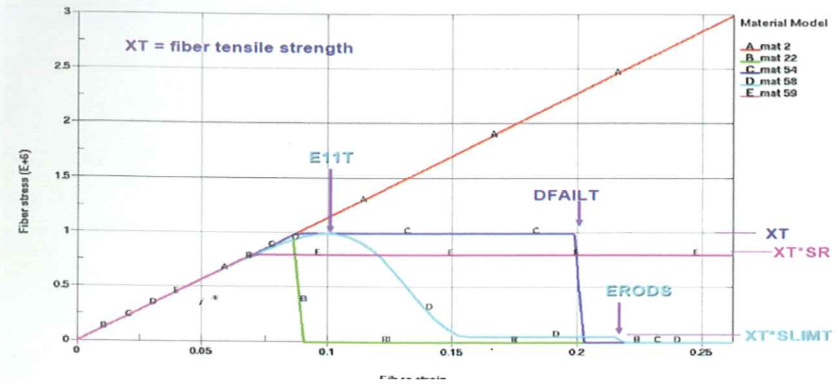


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between accuracy of results and simplicity of the model, as it needs not too much parameters in input. The option LAMSHT has been activated in order to account for a more precise transverse shear treatment, thus correcting the classical Reissner-Mindlin model associated with the FE adopted, i.e. a Belytschko-Tsai shell.

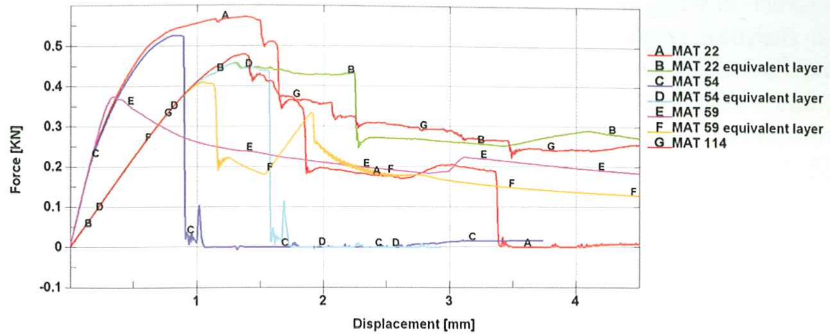


*Figure 4 - Force-displacement curves obtained with various material models and modeling choices for the composite skins*

## The model for the core, with virtual testing

In order to minimize the computation time for the largest problems, simplified models are usually necessary. This is the case for sandwich structures with honeycomb cores, which are made of a dense grid of small plates whose detailed modelling would be computationally very expensive, especially in an explicit analysis where the minimum dimension of the elements determines the numerical stability and time step of the whole model.

It is therefore important to use a simplified model for the core. There are many equivalent material models for these cores in LS-DYNA, to be used with solid elements. Among them there are MAT 63 (crushable foam), MAT 26 (honeycomb), MAT 126 (modified honeycomb), and MAT 142 (transversely anisotropic foam). All these materials require equivalent non-linear curves to define the constitutive equations under pure loads (e.g. pure shear or pure axial load), and can implement other parameters such as failure criteria and dumping for a more accurate simulation.
The chosen model is MAT 142 because it is relatively simple, requires few input curves and does not suffer from non-physical deformations due to the decoupling of curves, as encountered for example in MAT 26 and 126. This choice does not lead to any loss in accuracy of the core, since square cells make the core transversely isotropic on the macroscale.

| Symbol | Parameter | Insertedvalue |
|---|---|---|
| $\rho$ | Density | $187.4 \cdot 10^{-9}$ |
| $E_{11}$ | Axialelasticmodulus | 4.40 |
| $E_{22}=E_{33}$ | Transverseelasticmodulus | 1.14 |
| $E_{12}=E_{31}$ | Shear modulus in the axial direction | 0.894 |
| $E_{23}$ | Shear modulus in the transverse plane | 0.074 |
| $l_{ij}$ | The four load curve | See fig. 5 |
| $G$ | Shear modulus of base material | 39 |
| $K = \frac{E}{3(1-2\nu)}$ | Bulk modulus of the base material | 137 |
| $\mu$ | Internalviscositycoefficient | 0.075 |

*Table 2 - Input data inserted in MAT 142*

The material data used for the core are shown in table 2.
For such material models to be accurate, required curve data have to be gathered, usually through experimental tests. This kind of data is difficult to be found in literature, due to the variety of thicknesses, geometries and materials of the cores. Therefore dedicated experimental tests are needed for every different analysis, but can be expensive and time consuming, as they require dedicated machinery and enough samples to provide statistically significant correlations.

A more efficient way to obtain these data is the use of virtual testing which mimics the experimental tests. It has the added advantage of allowing many more tests to be performed so that parametric study/ optimization can be done on the core geometry and material. It must be emphasized that FEM is a very useful tool when some equivalent property is sought but no experimental data is available.

All simulations of the core have been performed using the implicit LS-DYNA solver. This solver is more appropriate for these non-linear quasi-static simulations, because it allows a solution without the need of searching for dummy methods to eliminate the inertial terms that are intrinsic in an explicit simulation.
Four equivalent curves are required for material MAT 142 and require a simulation for each one. The tests are: axial crush, lateral compression, axial shear and lateral shear, summarized in figure 5 along with their resulting curves. The curves, that are initially obtained in function of forces and displacement, have been manipulated to obtain the correct curve values in terms of stress and strain required in input by the material model MAT 142. The model for the later three tests is the same, with the only difference being the boundary conditions.

For the axial crush test a more refined model is required, as it is a post-buckling analysis, which is more difficult to be predicted. Therefore this model has a more refined mesh than the others, and uses the arc-length method. Furthermore, the initial imperfections in this model are simulated as a linear combination of the first two buckling modes of the model: so a linear buckling simulation was performed first. Two magnitudes of the maximum displacement for the first buckling mode have been taken into account: the first being one tenth of the cell wall thickness, and the second being one half. The maximum magnitude of the second buckling mode has been estimated as proportional to the ratio of the second to first critical buckling load. The two resulting curves do not show significant differences, as illustrated in Figure 5, so the one with

the smaller imperfection has been used.
In order to verify the reliability of the results, an analytical comparison can be made using the formulas in [Master I.G., Evans K.E. (1996), Models for the elastic deformation of honeycombs, Journal of Composite Structures 35, 403-422, Elsevier.]. For the shear modulus, the analytical formula is corrected by considering the shear stiffness of the cells. For the axial elastic modulus the effective area of material in pure axial compression is considered. The results of the comparison are shown in table 3.

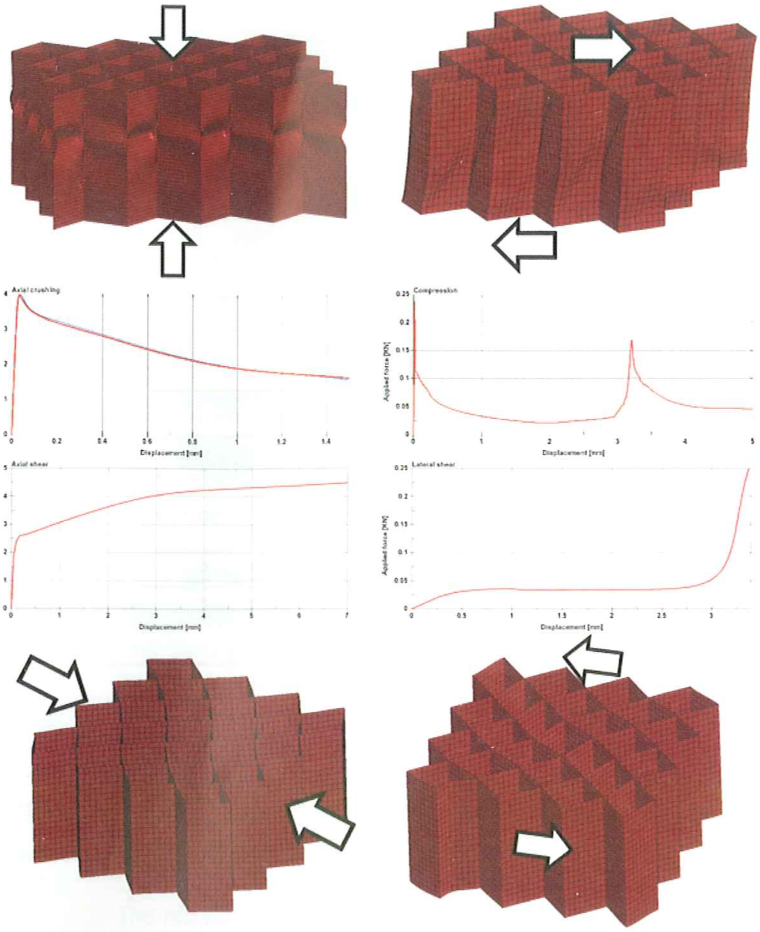

Figure 5 - Virtual testing of the cellular core, and resulting equivalent curves

### Experimental tests

Experimental tests have been performed on three samples in the flight vehicle structures laboratory at the Politecnico di Torino. Three-point quasi-static bending tests have been used for these tests. The dimensions of the samples agree with the specifications of the ASTM C 393-00 standard. The testing machine used is a Comazzi 20T, where the sample rests on two steel cylinders, 140mm apart from each other. The load is applied through a third cylinder with a diameter of 25 mm, put at the middle span of the laminate. The load is applied manually and the displacement at the middle cylinder is measured by two LVDT displacement transducers, situated on the two extremities of the section at the middle span of the laminate. Their values are then averaged in order to gain a more precise value of the displacement at the middle section. In order to obtain further information from the tests, a strain gauge was placed on the center of the compressed face of each sandwich laminate. The strain gauges have then been linked to a Wheatstone bridge to obtain a measure of the strain.

The environment during the tests was monitored; the room temperature was 22° Celsius with a humidity of 52%. The samples were deflected by about 4 mm, which were the maximum stroke of the displacement transducers. However, this deformation was sufficient to produce a failure mode.

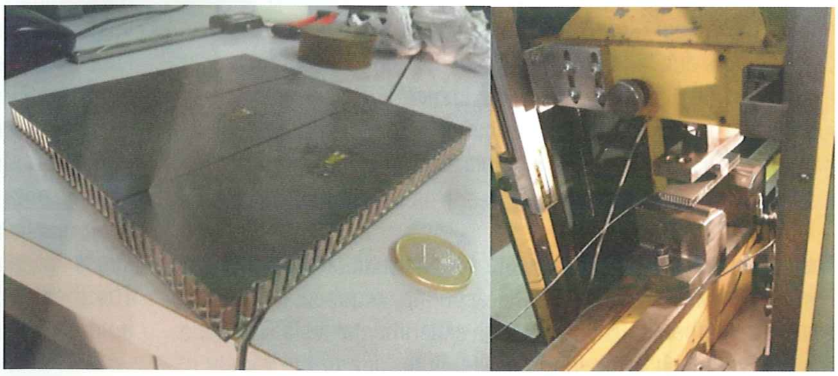
Figure 6 - Samples of the tested laminates[Tazzini A. (2014), Certification by analysis of aircraft seats: a numerical and experimental analysis to evaluate a composite made structure and a thoracic safety belt, master thesis discussed at Politecnico di Torino technical university, supervisors: Di Sciuva M., Maggiore P., Mattone M.]

From these tests the force-displacement curves of the laminates have been obtained; they are shown in figures 7 and 8.

From the experimental test results, it is possible to extract other interesting data to compare to the numerical model. This is possible using some hand calculations based on the Timoshenko beam theory, modified for a sandwich beam according to the ASTM standard.
In fact, for a sandwich beam loaded at the mid-point in 3 point flexure, it is possible to obtain the equivalent shear modulus G of the core as

$$G = \frac{4c}{(d+c)^2 b} U$$

where d is the total thickness of the sandwich beam, c is the thickness of only the core, b is the width, and U is the shear stiffness of the whole laminate. This can in turn be represented as the smaller imperfection has been used.

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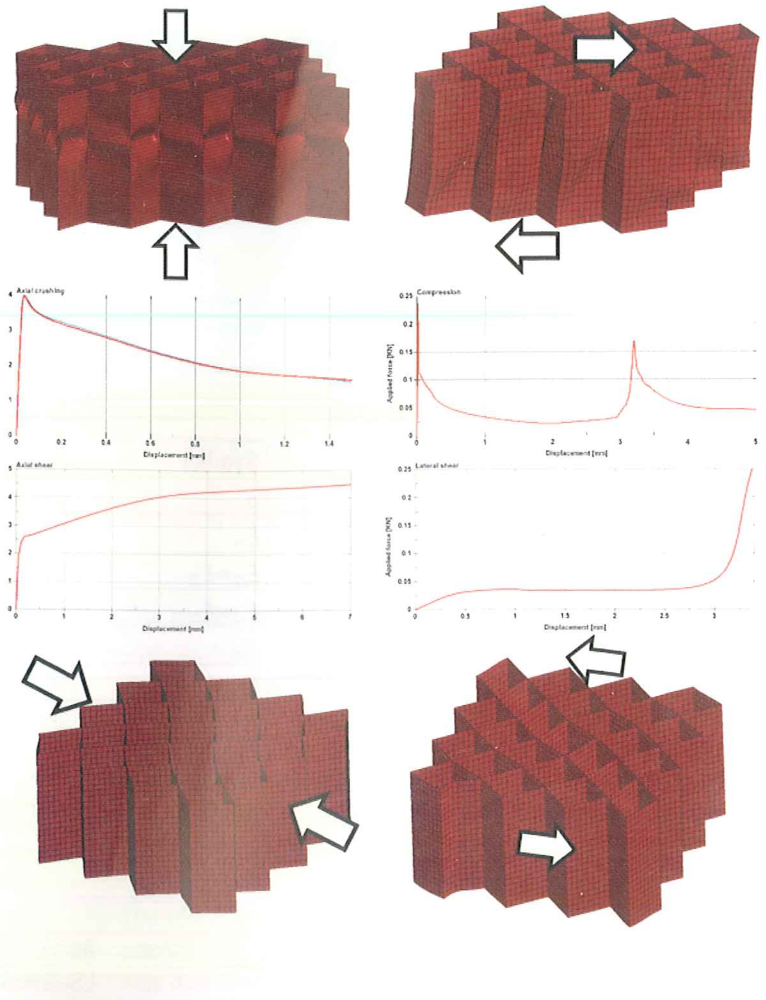
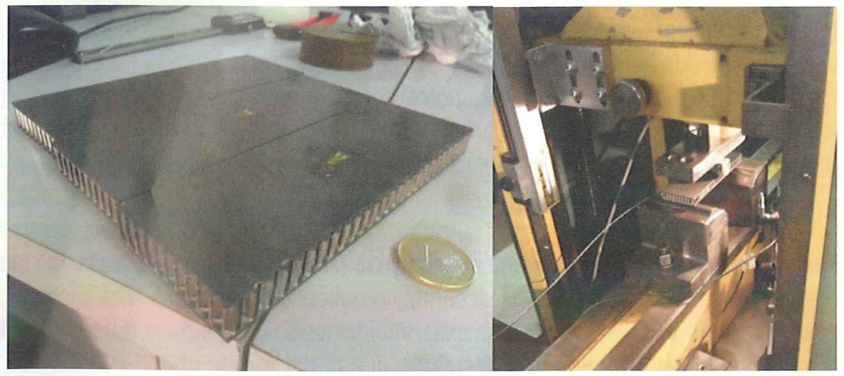


*Figure 5 - Virtual testing of the cellular core, and resulting equivalent curves*



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where d is the total thickness of the sandwich beam, c is the thickness of only the core, b is the width, and U is the shear stiffness of the whole laminate. This can in turn be represented as



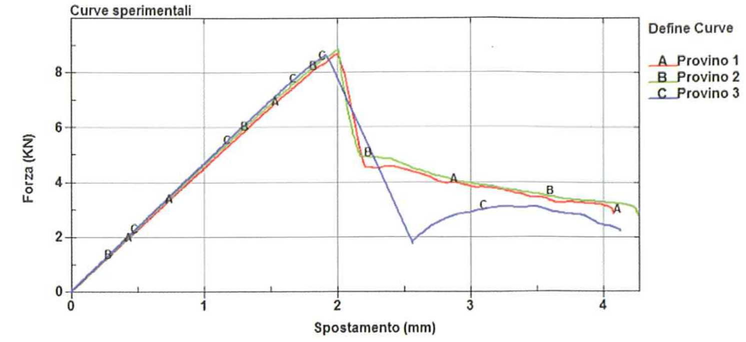


*Figure7 -Experimental force-displacement curves of the three samples, obtained from 3 point bending test [Tazzini A. (2014), Certification by analysis of aircraft seats: a numerical and experimental analysis to evaluate a composite made structure and a thoracic safety belt, master thesis discussed at Politecnico di Torino technical university, supervisors: Di Sciuva M., Maggiore P., Mattone M.]*

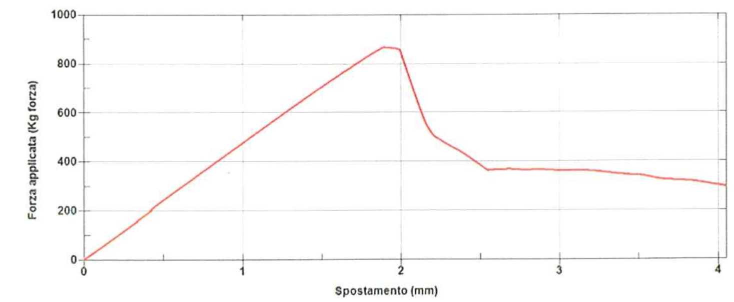


*Figure 8 - Averaged experimental force-displacement curves [Tazzini A. (2014), Certification by analysis of aircraft seats: a numerical and experimental analysis to evaluate a composite made structure and a thoracic safety belt, master thesis discussed at Politecnico di Torino technical university, supervisors: Di Sciuva M., Maggiore P., Mattone M.]*

$$U = \frac{PL}{4}\left(\frac{1}{W - \frac{PL^3}{48D}}\right)$$

where W is the maximum displacement on the middle point, P is the applied load, L is the useful length, and D is the flexural stiffness. This last term could not have been obtained by direct calculation from the data using the elastic moduli E of the faces, because these were not available from the manufacturer. So it has been obtained using the data gathered by the strain gauge, because from assumptions it follows that

$$M = D\frac{\partial\Phi}{\partial x}$$

with

$$\frac{\partial\Phi}{\partial x} = \frac{2\varepsilon}{d}$$

so that

$$D = \frac{M}{2\varepsilon}d$$

having called Φ the angle of rotation of the section. The strains ε longitudinal to the fibers where averaged between only two samples, because of problems on the data measuring of the third strain gauge; the value M/ε was averaged between all the values obtained in the proportional elastic range of each test.

## Evaluation of the mathematical model and recalibration

After obtaining the experimental data, it was possible to gain some of the mechanical properties for both the skins and the core.

For the skins it turned out that the equivalent elastic modulus is about $E_{eq}$=94.2 GPa, following the formula from ASTM

$$D = \frac{E(d^3 - c^3)b}{12}$$

which in turn gives an elastic modulus of 161 GPa for the fiber direction, and 10.7 GPa for the matrix direction, having assumed that their ratio is still 15, as found in literature. These values do not differ too much from the previously discovered data, having a difference of about 10%.

From experimental evidence and an iterative approach in FE model, it was found that the stress-strain curve of titanium taken from literature was of inferior mechanical properties when compared to the actual material of the core, and was therefore consequently recalibrated.

If actual material properties had been available, the final curve would probably fit better the experimental curves.

## Conclusions

The non-linear flexural behavior of a composite sandwich laminate has been investigated, so that the energy absorbed by the structure could be evaluated, by integrating the resulting force-displacement curve.

The force-displacement curve from the FE model set with LS-DYNA shows a good agreement with experimental data (figure 11). The biggest source of error is the lack of mechanical data of the original materials.

Table 4 summarizes the results of the experimental data and compares them with data from the FE model.

Analytical results of the cellular core properties are also added for comparison.

| | $P_{max}$ [KN] | D [KN·mm²] | U [KN] | G [GPa] | M/ε |
|---|---|---|---|---|---|
| **Average value** | 8.75 | 370 ·10³ | 573.6 | 0.891 | 58.3 ·10⁻³ |
| **Standard deviation σ** | 0.089 | 16.6 ·10³ | 85.7 | 0.137 | 2.6 ·10⁻³ |
| **Percentage σ** | 1.02 % | 4.49 % | 14.94 % | 15.38 % | 4.46 % |
| **Number of samples** | 3 | 2 | 2 | 2 | 2 |

*Table 3 - Data extrapolated from tests, with some statistically meaningful values*

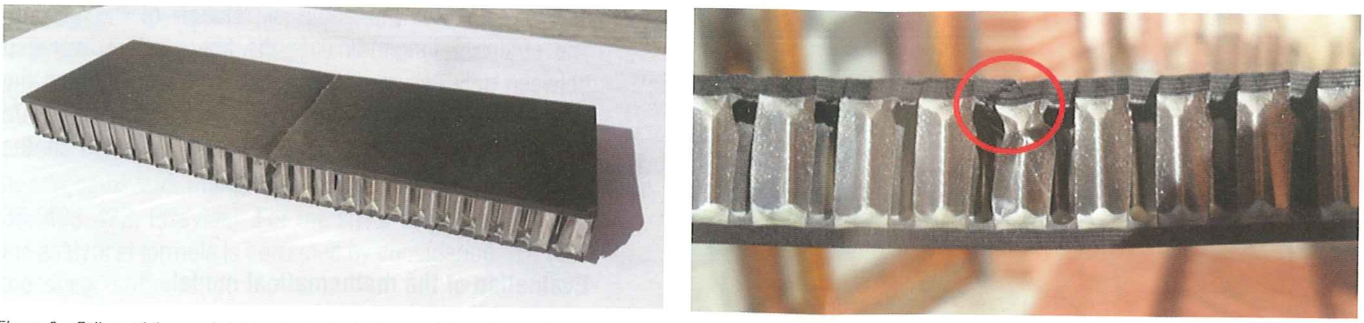

Figure 9 - Failure of the sandwich laminate; the failure mode is evidenced

| | Analytical | FE model | Experimental |
|---|---|---|---|
| E11 [Gpa] | 4.56 | 4.4 | - |
| E22=E33 [Gpa] | 0.057 | 1.14 | - |
| G12=G13 [Gpa] | 0.75-1.44 | 0.894 | 0.891 |
| G23 [Gpa] | 0.74-1.40 | 0.0740 | - |

Table 4 - Summary and comparison of results

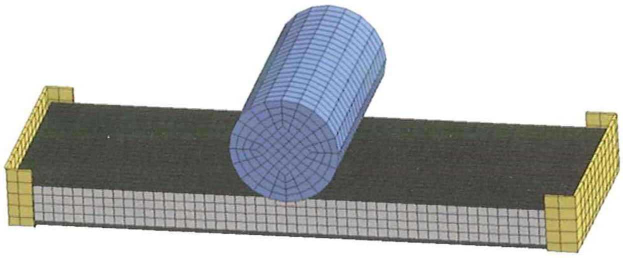

Figure 10 - FE model of the 3 point bending test

*Itacae is a consulting company from Turin that collaborates with EnginSoft in many projects regarding explicit dynamics using LS-DYNA code. The company has staff with extensive experience in the best-in-class methodologies for the analysis of industrial products & manufacturing processes, for the optimum management of the product development cycle. The main customers are manufacturers & design offices in the sectors of automotive & aerospace industry.*

### Application to an aircraft seat

The laminate in question has been applied to an aircraft seat, in order to investigate possibility of reducing the seat's mass, while improving its structural performance in crashworthiness.

## Using this advanced material allows the weight of the seat back to be reduced from 2.8 Kg to 1.4 Kg

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*Politecnico di Torino*

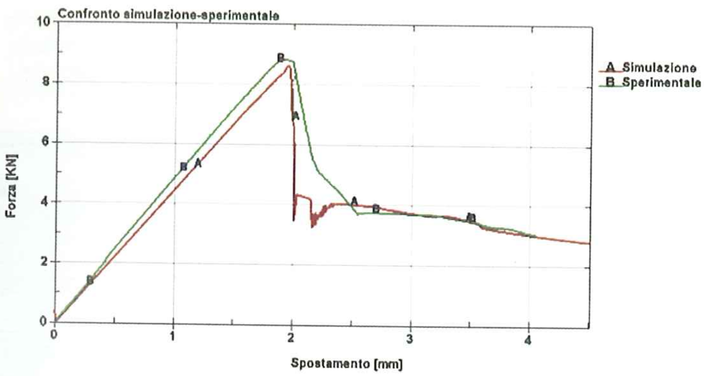


Figure 12 - Force-displacement plot of the FE model

Figure 11 - Crash test on the aircraft seat, modified using the studied composite laminate